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PETRONAS

**VIBRATION ANALYSIS OF DRILLSTRING WITH MASS IMBALANCE
USING FINITE ELEMENT METHOD**

By

Arun Kumar Shanmugam

Dissertation submitted in partial fulfilment of
the requirement for the
MSc. Petroleum Engineering

8th JULY 2013

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CERTIFICATE OF APPROVAL

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Approved by,

Dr Sonny Irawan

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TRONOH, PERAK

JULY 2013

CERTIFICATION OF ORIGINALITY

This is to certify that I was responsible for the work submitted in this project, that the original work is my own except as specified in the reference and acknowledgements, and that the original work contained herein have not been undertaken or done by unspecified sources or persons.

Arun Kumar Shanmugam

ABSTRACT

Drill string vibrations pose a major challenge to the operational efficiency of the drilling process in the oil and gas industry. The mass imbalance in the drill string caused by MWD tools and drill collar sag disrupts the accuracy of the drilling operation and can give rise to huge vibration thereby damaging the down hole components. Since the surface readings are not reliable measurements of drill string lateral vibration, it becomes necessary to study drill string vibration under Finite Element method. In this project, ANSYS software was used to build the geometry of a drill string model and appropriate boundary conditions were applied to the model such that the drill string experiences lateral vibrations. The component of mass imbalance was introduced to the drill string by adding an equivalent weight eccentrically on one side of the drill string. Through the use of ANSYS simulation modal and harmonic analysis were performed to study the vibration characteristics of the drill string. Modal analysis was performed to determine the mode shapes and the first three critical frequencies of the drill string. The effect of mass imbalance was studied by comparing the first three critical frequencies before and after the addition of the imbalance. The results showed an increase in natural frequency to 2.03Hz, 2.23Hz, and 4.63Hz. The analysis was repeated with different configurations of drill string to understand the effect of length on the natural frequencies. Harmonic analysis was performed on the drill collar with mass imbalance to study the frequency response to lateral displacements. The results of this project can be used to identify the safe operating ranges of rotary speed for the drill string and identify the lateral displacement of a critical component for a range of frequencies, thereby avoiding damage to the drill string.

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TABLE OF CONTENTS

CHAPTER 1: INTRODUCTION	1
1.1 Background of study	1
1.1.1 Rotary system.....	1
1.1.2 Drill string	2
1.1.3 Drill pipe	3
1.1.4 Heavy weight drill pipes	3
1.1.5 Drill collars.....	3
1.1.6 Stabilisers	3
1.1.7 Drill bit.....	4
1.2 Problem statement.....	5
1.3 Objective and scope of study	5
CHAPTER 2: LITERATURE REVIEW	6
2.1 Effects of drill string vibrations	6
2.2 Modelling of drill strings	8
2.3 Analysis of the Model	10
CHAPTER 3: METHODOLOGY	13
3.1 Importance of Lateral vibration	13
3.2 Drill string Mass imbalance	13
3.2 Finite element analysis of drill string.....	15
3.3 Drill string modelling.....	16
3.3.1 Geometric modelling.....	16
3.3.2 Drill collar imbalance.....	19
3.3.3 Materials and methodology.....	20
3.3.4 Boundary conditions	20
3.4 Modal analysis	23
3.5 Harmonic analysis.....	24
CHAPTER 4: RESULT AND DISCUSSION	26
4.1 Modal analysis	26
4.2 Harmonic analysis.....	32

CHAPTER 5: CONCLUSION AND RECOMMENDATION	35
5.1 Conclusion	35
5.2 Recommendation.....	36
REFERENCE.....	37
APPENDIX	38

LIST OF FIGURES

Figure 1: Drill String components	2
Figure 2: Types of vibrations	6
Figure 3: Lateral vibration mode shapes	8
Figure 4: Static deflection and eigen modes	9
Figure 5: Stick slip whirl model.....	10
Figure 6: Mass imbalance	13
Figure 7: Modeling with ANSYS	15
Figure 8: Drill string design	18
Figure 9: Mass imbalance in drill string	19
Figure 10: Cylindrical support at bit	21
Figure 11: Cylindrical support at stabilizer.....	21
Figure 12: Cylindrical support at the Rotary table.....	22
Figure 13: Harmonic analysis	25
Figure 14: First deformation Drill string 1	28
Figure 15: Second deformation Drill string 1	28
Figure 16: Third deformation Drill string 1	29
Figure 17: First deformation Drill string 1 with mass imbalance	29
Figure 18: Second deformation Drill string 1 with mass imbalance.....	30
Figure 19: Third deformation Drill string 1 with mass imbalance.....	30
Figure 20: Harmonic analysis of drill string 1	32
Figure 21: Harmonic analysis of drill string 2	33
Figure 22: Harmonic analysis of drill string 3	33

LIST OF TABLES

Table 1: BHA configuration used in field and experimental study	11
Table 2: Field experiment result of first three critical frequencies	12
Table 3: Base case drill string configuration 1	16
Table 4: Drill string configuration 2	17
Table 5: Drill string configuration 3	17
Table 6: Properties of Drill string and drilling mud.....	18
Table 7: Comparison of ANSYS result with Burgess result.....	26
Table 8: Natural frequencies of drill string under different conditions	26

NOMENCLATURE

Symbol	Expansion	Unit
M	Mass	Kg
K	Stiffness	N/m
δ	depth of penetration	m
ω	Angular velocity	Rad/s
r	Deflection of shaft	m
e	eccentricity	m
[M]	Mass matrix	
[K]	Stiffness matrix	
[I]	Unity matrix	
u	Velocity	
\ddot{u}	Acceleration	

CHAPTER 1

INTRODUCTION

1.1 Background of study

In the field of oil and gas industry, wells are drilled through the reservoir to produce the recoverable hydrocarbons to the surface. The drill bits are the cutting tools which are used to drill these oil wells. These drill bits are rotated around using a series of tubular pipes, collectively known as the drill string, which can be thousand meters long. In order to turn the bit around, the entire drill string is rotated from the surface using the rotary table. The drill string is suspended from the swivel and Kelly, therefore the upper part of the drill pipe section is in constant tension while the lower part or the BHA is resting on the rock formation thus under compression.

The hollow drill string acts as a conduit through which drilling mud is pumped as it enters the annulus between the drill string and borehole wall through the bit nozzle. The primary function of the mud is to prevent the formation fluid from entering the well and to transport back the cuttings to the surface.

The detailed description of the components are given below

1.1.1 Rotary system

The rotary system is used to rotate the drill string thereby transmitting rotation to the drill bit. The rotary system includes all the components required to achieve rotation. The swivel is at the top of the drill string which is capable of

- Supporting the drill string
- Allowing the drill string to rotate
- Allowing the drilling mud to be pumped while the drill string is in rotation

The first section of pipe below the swivel is called the Kelly. It is about 40 feet long and has a hexagonal structure to transmit the rotary motion from the rotary table to the drill string. The rotary table is located on the drill floor and can be turned in both clockwise and anti clockwise direction from the driller's console. The torque from the rotary table is transmitted to the Kelly through the four pins on a device which

runs along the length of the Kelly called the Kelly bushing. Rotary motion is supplied to the Kelly from the rotary table through the rotary bushing.

1.1.2 Drill string

Drill string is used to describe the tubulars and accessories on which the drill bit is run to the bottom of the borehole. The drill string consists of drill pipe, drill collars, the Kelly and various other pieces of equipments such as stabilizers, reamers, which are included in the drill string just above the drill bit as shown in Figure 1. The equipments at the bottom of the drill string are collectively called the Bottom Hole Assembly or the BHA. The main function of the drill string is

- To suspend the bit
- To transmit rotary torque from the Kelly to the bit
- To provide a conduit for circulating drilling fluid to the bit

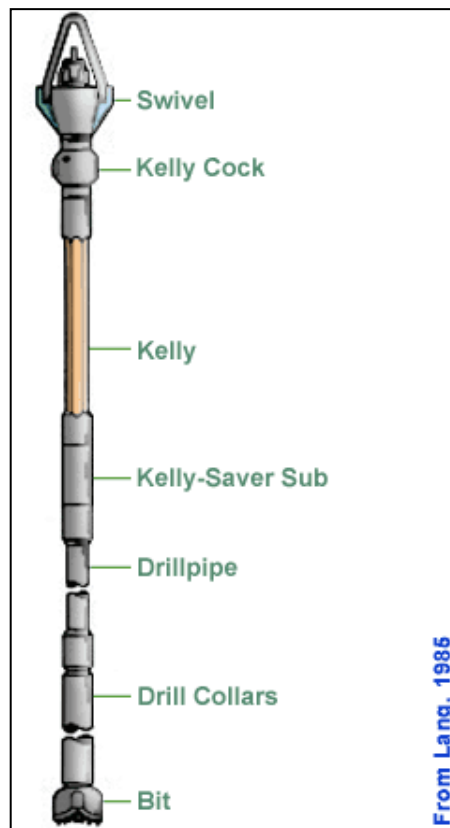


Figure 1: Drill String components (Lang, 1986)

1.1.3 Drill pipe

The drill pipe is the major component of the drill string. It is a seamless pipe with threaded connections, known as tool joints. One end of the drill pipe consists of the box, or the female end and the other end has pin or the male end. Naturally the wall thickness of the tool joint is greater than that of the drill pipe. The drill pipes are added along as required by the drilling programme.

1.1.4 Heavy weight drill pipes

Heavy weight drill pipes or HWDP are similar to the regular drill pipes but with increased wall thickness. The weight of the HWDP is thus higher than that of a drill pipe. It is usually placed in the transition between drill pipes and drill collars. They usually have a spiral structure to prevent pipe sticking.

1.1.5 Drill collars

The drill collars are tubular which have a much larger outer diameter and generally smaller inner diameter than drill pipe. The main functions of which is

- To provide enough weight on bit for efficient drilling
- To keep the drill string in tension, thereby reducing bending stresses and failures
- To provide stiffness in the BHA for directional control (HWU 2005).

1.1.6 Stabilisers

Stabiliser is a length of pipe which has blades on its outer surface. These blades may be straight or spiral whose function is to

- Reduce buckling and bending stresses on the drill collar
- Allow higher weight on bit
- Prevent wall sticking
- Increase the bit life by reducing wobble

1.1.7 Drill bit

A drill bit is the cutting or boring tool which is found at the end of the drill string. The bits drill through the formation by scraping, chipping, gouging or grinding the rock at the bottom of the hole. The drilling bit has a provision through which drilling mud is circulated to remove the drilled cuttings.

1.2 Problem statement

As the drill string moves down hole, it is subjected to a variety of stresses, including tension, compression, vibration, torsion, friction, formation pressure and circulating fluid pressure. All of which will lead to the failure of the drill string if not monitored properly. These vibrations in particular may disrupt the accuracy of the drilling operations and can cause damage due to fatigue. Mass imbalance on drill string has been found to be an important source of lateral vibration. Therefore it is vital to carry out vibration analysis, to understand its effects and develop ways of preventing it.

Through this project, Finite element method is used to understand the vibration of a typical drill string subject to mass imbalance.

1.3 Objective and scope of study

The main aim of this project is

- To model a drill string using finite element method software such as ANSYS
- To include the component of mass imbalance in the model and perform modal and harmonic analysis
- To perform modal analysis on the aforementioned model to find out the first three critical frequency, critical speeds and mode shapes
- To perform harmonic analysis on the critical component to understand the frequency response to lateral displacement
- To compare and interpret the result of both the analysis to understand the effect of mass imbalance on the drill string

CHAPTER 2

LITERATURE REVIEW

2.1 Effects of drill string vibrations

According to (Schlumberger, 2010) drill string vibrations can be divided into three types- axial, lateral and torsional as shown in Figure 2. ***Axial vibrations*** can cause bit bounce which can damage bit cutters and bearings. ***Torsional vibrations*** can cause irregular down hole rotation. Stick/slip is often seen while drilling and is a severe form of drill string torsional oscillations in which the bit becomes stationary for a period. As the severity of stick slip increases, the length of the stuck period increases, as do the rotational accelerations as the bit breaks free. Torsional fluctuations fatigue drill collar connections and can cause damage to the bit. ***Lateral vibrations*** are the most destructive type of vibration and can create large shocks as the BHA impacts the wellbore wall. The interaction between BHA and drill string can cause backward whirl sometimes, which is a severe form of vibration, creating a high-frequency large magnitude bending moment fluctuations that result in high rates of component fatigue.

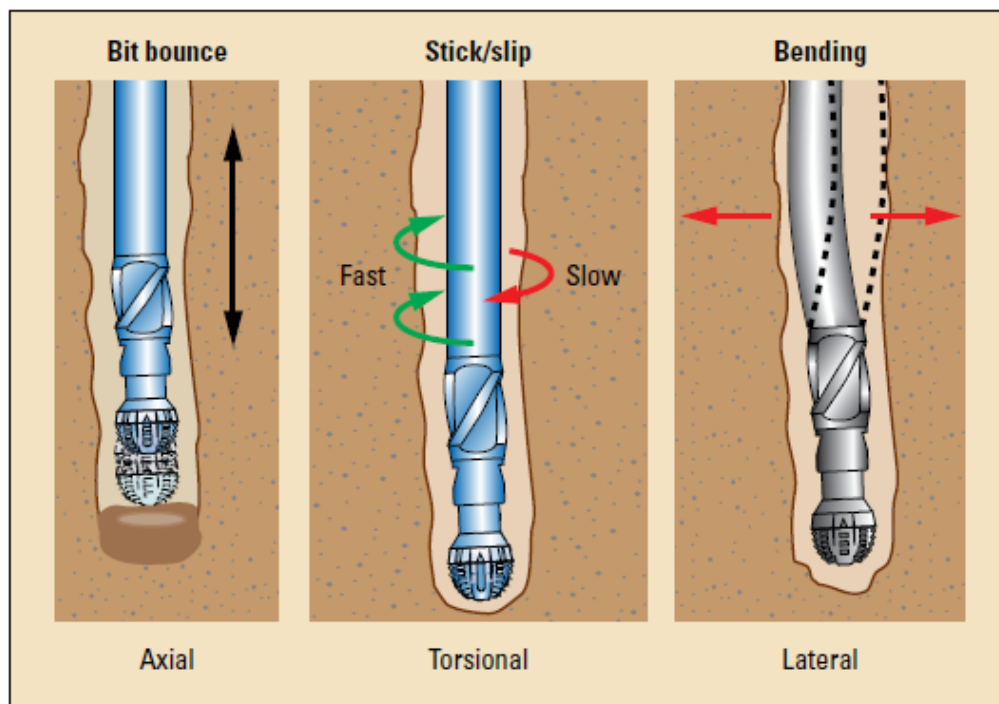


Figure 2: Types of vibrations (Schlumberger, 2010)

There is a wide range of potential excitation sources such as mass imbalance, misalignment and kinks or bends, the cutting action of drill bit, stabilizer blades, mud motors and the friction factor between borehole and drill string.

(Harvey and Wassel, 1991) published a paper on the effects of mud motor imbalance as a source of drill string vibrations. High torque/ low speed motors have rotor orbiting eccentrically about the centre of the motor. This imparts large dynamic loads to the BHA. Finite element analysis of the proposed BHAs consisting of two motor configurations was used to determine the critical speeds, as well as the speeds that cause the stabilizers to bounce from side to side.

(Dykstra, 1995) published a paper which states that mass imbalance is a major source of down hole lateral vibrations. Factors that contribute imbalance include bore misalignment, initial curvature, and gradual wear during service. Field experiments conducted by him focused on drill collars, as they are a common source of imbalance. He had neglected the effect of drilling bit by replacing it with a bull nose. Lateral vibrations caused by the collar/ wellbore collision were measured at various locations on the drill string using sensors.

(Costa and Ribeiro, 1997) published a paper on the mechanical behaviour of unbalanced drill collars by removing 120 kg eccentricity from a standard 30 feet drill collar. Stabilizers cause significant increase in torque and drag in drill string. The main feature of this paper was that it investigated the effect of mass unbalance of drill collars as means of mitigating the effect of drill string inclination without the use of stabilizers. The unbalance in the drill string was created by machining a sequence of holes along the drill collars

(Jamal and Seyed, 2011) conducted finite element analysis of a drill string for lateral vibrations in deviated wells, which took account of the contact points between drill string and bore hole as well as the effect of mud fluid. A static preliminary analysis was done to determine the effect of length subject to lateral vibrations. Later modal, harmonic and transient analysis was done to determine the modes, natural frequencies, frequency response and time dependent response

2.2 Modelling of drill strings

According to (Schlumberger, 2010) there are two main types of vibration models: frequency domain and time domain. In the frequency models, a static model is used to compute BHA touch points and this information is used to compute the natural frequencies of the drill string. The time domain models are more computationally intensive and typically, drill forward in time and model to the interaction between drill string and borehole wall. This allows the model to capture mechanisms, such as forward and backward whirl, and provide a better representation of the excitation because of rotation of bends and kinks and to mud motor rotation. Figure 3 shows the first three natural frequencies and mode shapes of lateral vibration.

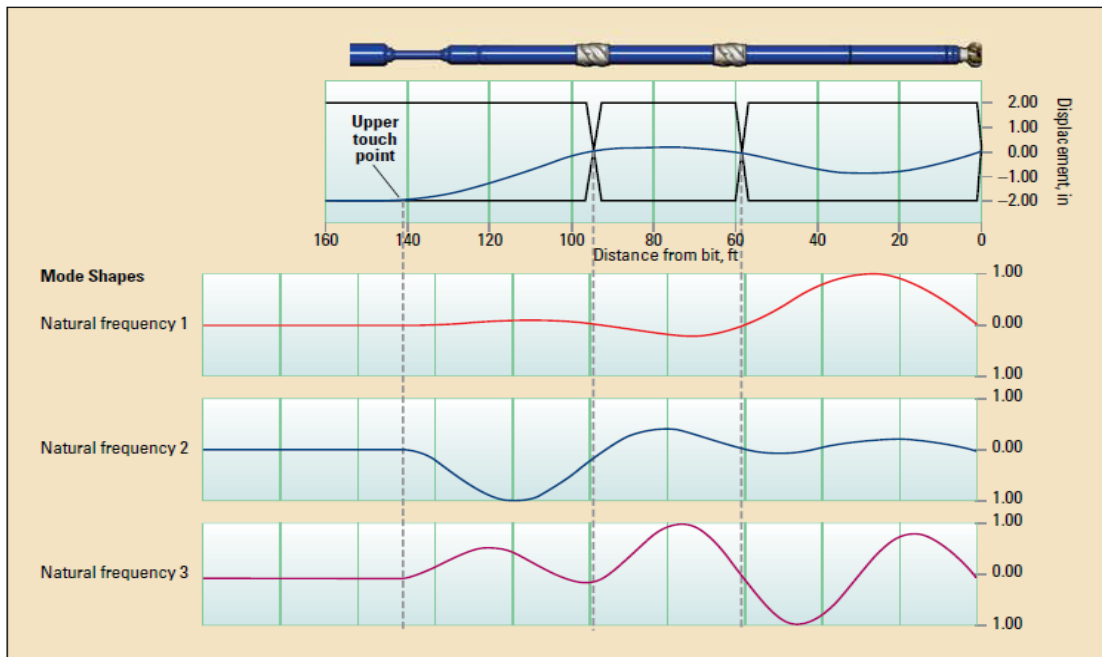


Figure 3: Lateral vibration mode shapes (Schlumberger, 2010)

The modelling was subject to limitations because it requires inputs that are commonly unknown or not available in real time, including formation properties and heterogeneity, BHA component imbalance and orientation, BHA misalignment (bent collars), down hole damping, friction factor at contact points. Models are also sensitive to hole diameter or stabilizer clearance.

Based on the study by (Kreuzer and Struck, 2003) vibrations in long torsional strings result in spatio-temporal dynamics. These vibrations disturb the accuracy of the drilling process and may cause damage due to fatigue. In their paper, a model for the

drill string is proposed which consists of analysis of bending vibrations, self-excited torsional vibrations as well as the coupling between them. The aim of the model with only few degrees of freedom was the development of an optimal control technique for the active damping of drill-string vibrations. The string was modelled as a one-dimensional continuum with a stress-free and a deformed configuration.

The simplified Lagrangian yields the equations of motion which were solved approximately using Galerkin's method. The external forces along the string comprised the load due to the interaction with the fluid as well as contact forces with the bore hole. The contact was treated as impact, as a geometric constraint, or by penalty functions.

Hence, a linear relationship is used:

$$FP = K \cdot \delta$$

Where K denotes stiffness and δ represents the depth of penetration.

The solution of which is illustrated in Figure 4:

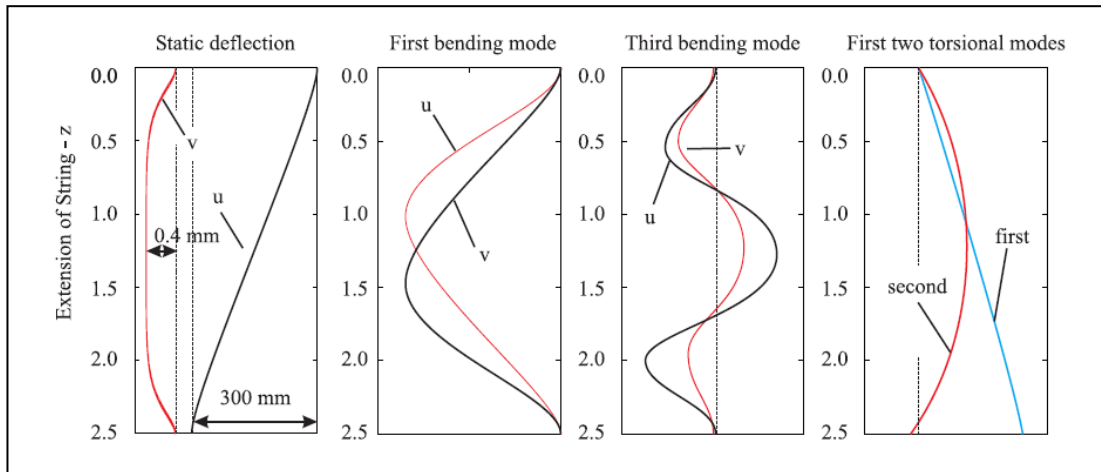


Figure 4: Static deflection and eigen modes (Kreuzer and Struck, 2003)

(Leine and van Campen, 2000) published a paper investigating the combined effects of torsional stick slip and lateral whirl vibration. It is observed that the complicated behaviour maybe due to the fluid forces in the drilling mud. A stick slip whirl model as presented by them in Figure 5 is a simplified drill string confined in borehole conditions with interactions with drilling mud. The simplest stick slip model that can

qualitatively describe the observations is a one-DOF model for the torsional vibration and a two DOF model for the lateral vibration. The disappearance of stick slip vibration when lateral whirl vibration appears is explained by bifurcation theory. Their theoretical results are compared with actual real time results from a drilling rig.

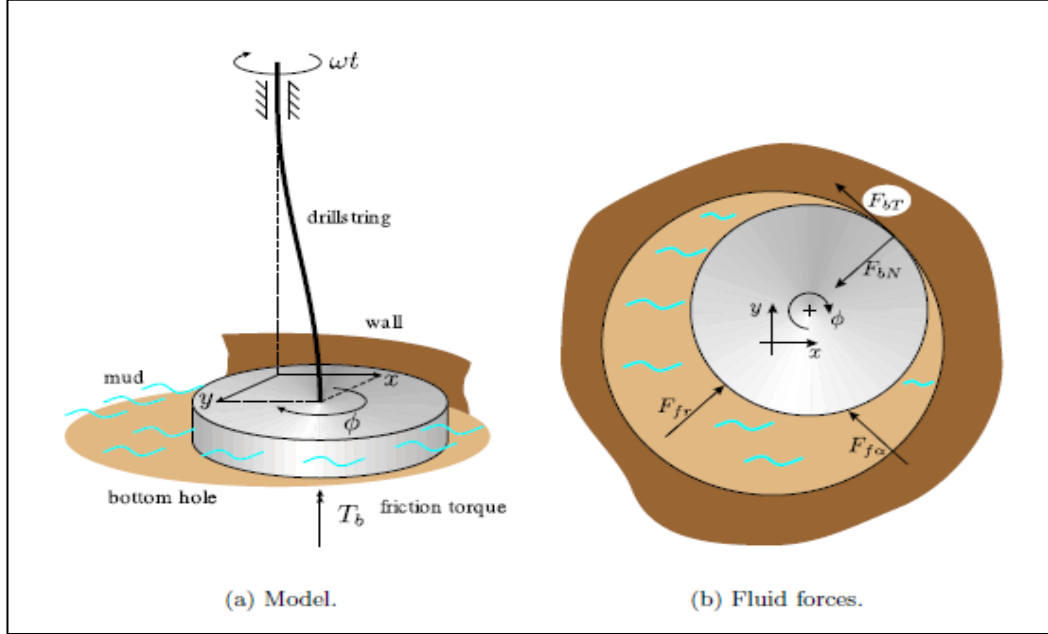


Figure 5: Stick slip whirl model (Leine and. van Campen, 2000)

2.3 Analysis of the Model

(Pasley et al, 1994) analyzed that the lateral vibrations of drill string bottom hole assemblies were confirmed by sensing longitudinal and torsional accelerations or displacements of the drill string at or near its top end during rotation and were able to determine the lateral critical speed and displacements of the drill string from the mode shape of the bottom hole assembly which occurs during forward synchronous whirl. Lateral displacement of the bottom hole assembly due to impacting of the borehole wall introduces a torsional impulse which, together with considerable lateral motion, is repeated each revolution. The lateral motion causes a small axial motion to develop in the bottom hole assembly which results in an axial force and axial shortening of the drill string which varies once per revolution.

(Dawson, Lin and Spanos, 1987) investigated torsional stick-slip oscillations in drill strings semi- analytically. Assuming a piecewise linear friction curve in the slip phase, they were able to apply this to a one degree of freedom model. When compared with experimental results it showed good agreement. The field measurements were obtained from a deep low-angle directional well. The theoretical results showed that amplitude and frequency increased with rotary speed. The model was also able to predict a critical rotary speed above which stick-slip oscillations would not occur, which was also seen in agreement with the field. They had finally concluded that this oscillation can be reduced by lowering the static friction coefficient.

(Burgess, Mc Daniel and Das, 1987) published a paper which includes theoretical and FEM vibration studies on several drill string BHA designs which were used in the field. These BHAs were prone to failure due to resonance. The field example that was used in the study had the following configuration:

Table 1: BHA configuration used in field and experimental study (Burgess, Mc Daniel and Das, 1987)

Element number	Name	Length (feet)	O.D (inch)	I.D (inch)	Blade O.D	Cumulative length (feet)
1	Bit	3.25	6.25			3.3
2	Stabilizer	6.40	4.75	2.25	6.25	9.7
3	D.C	30.8	4.75	2.25		40.5
4	stabilizer	6.60	4.75	2.25	6.25	47
5	D.C	529.1	4.75	2.25		576.2
6	HWDP	121.9	3.50	2.06		698.1

And the result for first three critical frequencies:

Table 2: Field experiment result of first three critical frequencies (Burgess, Mc Daniel and Das, 1987)

	length	1 st natural frequency (Hz)	2 nd natural frequency(Hz)	3 rd natural frequency(Hz)
Burges et al(1987)	114(ft)	1.23	3.15	4.19

In conclusion from the literature survey it is understood that

- There are three types of vibrations commonly experienced in drill string- axial, lateral and torsional vibrations
- The causes of such vibrations should be taken into account while modelling the drill string
- There are two generalized types of modelling to study vibrations – Time and frequency modelling
- The theoretical drill string models were setup to be under one degree of freedom for simplicity but have included parameter variations such as a bending, multiple drill pipe
- Theoretical vibration analysis includes solving for complex equations of motions analytically
- The amplitude of stick-slip vibration increases as rotary speed increases, but the vibration effect is negated when the rotary speed passes the critical speed.
- Mud motor imbalance and drill string mass imbalance are important sources of lateral vibrations
- Field experiments have been conducted on a standard 30 feet drill string with mass imbalance of up to 120 Kg on one of its side
- Although finite element method has been used to study drill string vibrations, the effect of mass imbalance on drill string lateral vibrations have not been studied using Finite Element Modelling
- The drill string configuration used by (Burgess, Mc Daniel, Das) will be used for the study for validation purposes.

CHAPTER 3

METHODOLOGY

3.1 Importance of lateral vibration

Recent studies have suggested that drill string failures due to vibration have been encountered more commonly in vertical wells than in deviated wells. The reason for this is the use of stabilisers in deviated wells which have reduced the torsional and axial vibration vibrations significantly by damping it. The bit bouncing is an important source of axial and torsional vibrations. It is found that, the drill collar weight normal to the drill string axis and the wellbore wall effect are the stabilizing force in the lateral modes of vibrations

Lateral vibrations depend mostly on drill collar sizes and position of stabilizers. Generally it is found that the more the mass of drill string, the lesser the stiffness and optimum use of stabilizers the lesser the resonance frequencies observed.

3.2 Drill string mass imbalance

Lateral vibrations of drill string causes fatigue failures, excessive wear, Measurement While Drilling (MWD) tool failure. A rotating body is said to be unbalanced when its centre of mass doesn't coincide with the axis of rotation. When rotating a shaft with unevenly distributed mass, its centre of mass does not coincide with the axis of rotation, thus it causes vibrations as explained by Figure 6.

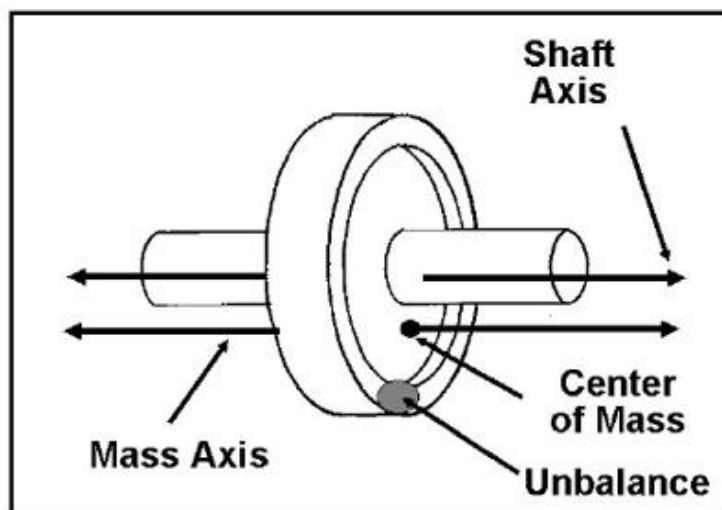


Figure 6: Mass imbalance (Dykstra, 1995)

The distance between the centre of gravity and rotational axis is termed as eccentricity e . This eccentricity causes a centrifugal force to act on the centre of gravity when the shaft is rotated. This unbalanced force causes the shaft to whirl. When the unbalanced shaft is rotated at one of its natural frequencies of lateral vibrations, the deflections become very large. This is termed as resonance and the speed at which resonance occurs is called critical speed.

Consider a weightless shaft with a mass M at the middle. Supposing the centre of mass is not at the centre line. When the shaft rotates, centrifugal forces cause it to bend out. Let the deflection of the shaft be r . The distance to the centre of gravity is then $r+e$. the shaft rotates at ω rad/s. the stiffness of the shaft is Kt N/m

Therefore the centrifugal force is $M\omega^2 (r+e)$

And the deflection force $F = Kt \cdot r$

Equating the force we have,

$Kt \cdot r = M\omega^2 (r+e)$ from which $r = M\omega^2 (r+e) / Kt$

$$r = \frac{M\omega^2 e}{Kt(1 - \frac{M\omega^2}{Kt})}. \text{ It has already been shown that } Kt/M = \omega_n^2$$

$$r = \frac{\omega^2}{\left(1 - \frac{\omega^2}{\omega_n^2}\right)\omega_n^2} e$$

$$r = \frac{e}{\left(\frac{\omega}{\omega_n}\right)^2 - 1}$$

This same principle applies to the rotation of the drill string. There are several sources of mass imbalance on the drill string such as bore misalignment, initial curvature of the drill string or gradual wear during service. A common source of imbalance occurs at the drill collars due to MWD tools.

3.2 Finite element analysis of drill string

Finite element modeling is a technique used for calculating approximate solutions to ordinary differential equations (ODE) or partial differential equations (PDE). It finds its applications either when the equations used or the geometry on which the solution required is complex. The method exchanges solving a single ODE or PDE on a larger complex domain for solving a much larger number of algebraic equations on simple shaped pieces that makes up the actual solution domain.

ANSYS is a finite element modeling package for computing solutions for static, dynamic structural analysis, heat transfer, fluid mechanics, acoustic and electromagnetic problem. In this project, modal analysis, harmonic analysis is used to determine the vibration characteristics of the drill string. The typical structure of the modeling process using ANSYS is shown in Figure 7.

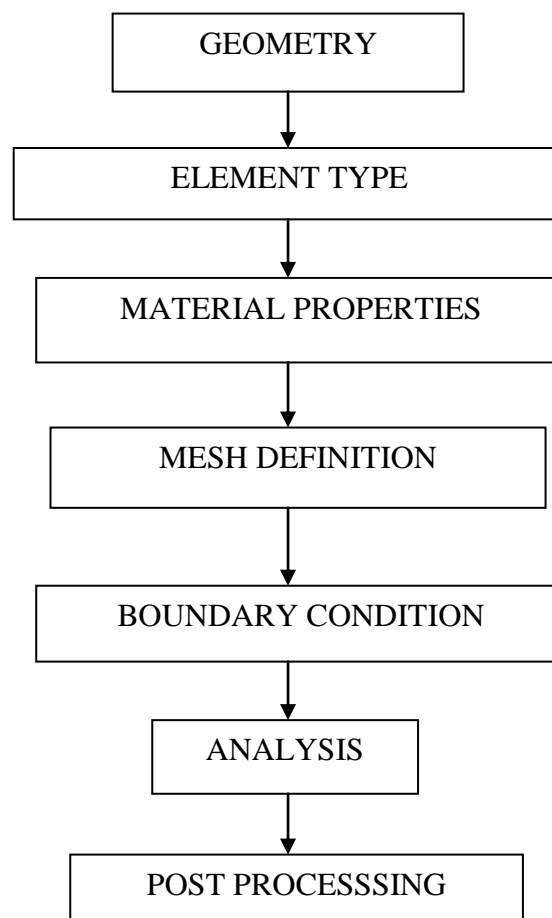


Figure 7: Modeling with ANSYS

3.3 Drill string modelling

A drill string of certain specification was used by Burgess in 1986 for his vibration analysis. The data for this drill string was obtained from a field experiment which was subject to failure due to resonance. The field experiment was such that the wellbore was inclined at .5 degree and the effective length of the drill string was around 700 feet as shown in Table 1. The first critical frequency was encountered around 3-3.4 Hz.

Burgess conducted a finite element analysis for the drill string configuration and found through static analysis that the mode shape curves tend to zero at 113 feet. This implies that the drill string subject to lateral vibration was only 113 feet which is taken as the cut off point.

Therefore in this project, the drill string is modeled up to 114 feet, thereby eliminating the effect of well deviation on the model. However, the wellbore effect is taken into account through the boundary conditions which will be specified subsequently.

3.3.1 Geometric modelling

To analyse the effect of different length and weight on bit on drill string vibration, 3 configurations of drill string have been put forward as follows

Table 3: Base case drill string configuration 1

Name	Length (ft)	O.D (in)	I.D (in)	Aggregate length (ft)
Bit	3.25	6.25		3.3
Stabilizer	6.40	4.75	2.25	9.7
Drill collar	30.8	4.75	2.25	40.5
Stabilizer	6.60	4.75	2.25	47.1
Drill collar	30	4.75	2.25	77.1
HWDP	30.42	3.50	2.06	107.5
Rotary table	3	3.50		110.5

Table 4: Drill string configuration 2

Name	Length (ft)	O.D (in)	I.D (in)	Aggregate length (ft)
Bit	3.25	6.25		3.3
Stabilizer	6.40	4.75	2.25	9.7
Drill collar	30.8	4.75	2.25	40.5
Stabilizer	6.60	4.75	2.25	47.1
Drill collar	60	4.75	2.25	107.1
HWDP	30.42	3.50	2.06	137.5
Rotary table	3	3.50		140.5

Table 5: Drill string configuration 3

Name	Length (ft)	O.D (in)	I.D (in)	Aggregate length (ft)
Bit	3.25	6.25		3.3
Stabilizer	6.40	4.75	2.25	9.7
Drill collar	30.8	4.75	2.25	40.5
Stabilizer	6.60	4.75	2.25	47.1
Drill collar	90	4.75	2.25	137.1
HWDP	30.42	3.50	2.06	167.5
Rotary table	3	3.50		170.5

Using ANSYS software, the user can build up the drill string with the bit, stabilisers, drill collars, heavy wall drill pipe and rotary table sequentially as follows

- Select mechanical model which will open onto a new window. On this window, geometry is selected which will allow geometric modelling.
- From the primitives menu, cylinders can be created with necessary dimensions
- The solid cylinder represents the equivalent of a bit without its complex designs

- Similarly the stabilizers with its specific outside and inside diameters are created
- The drill collar, Heavy wall drill pipe and rotary table are all created in a similar fashion

The following figure shows the geometry of the drill string design (not subject to scale) used for simulations

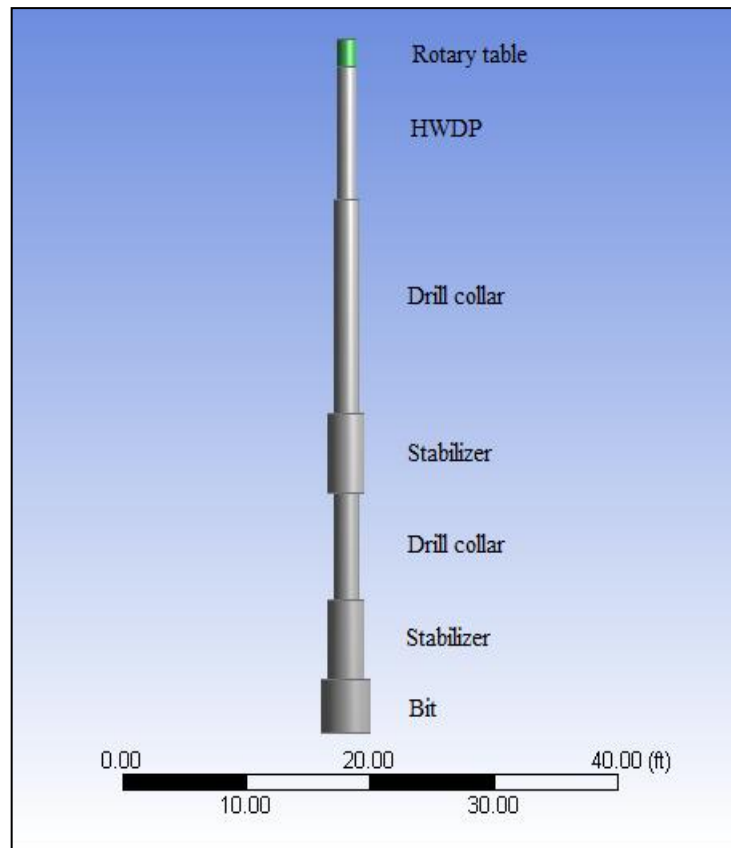


Figure 8: Drill string design

Since the mud effect will be considered in the analysis, the mechanical properties of drill string and mud are as follows:

Table 6: Properties of Drill string and drilling mud

Drill string	Modulus of elasticity(lb/ft ³)	4.32*10 ⁹
	Density (Slug/ft ³)	15.2
	Poisson ratio	.23
Drilling mud	Density (PPG)	12

3.3.2 Drill collar imbalance

The purpose of the project is to determine the effect of mass imbalance on the drill string vibrations. From literature survey, it was found that up to 120 Kg of material was removed eccentrically from one of its side in a standard 30 feet drill collar. This was primarily done to mitigate the effect of drill string inclination in a deviated well without the use of stabilizers.

Drill collars have been subject to mass imbalance due to MWD tools or initial curvature or gradual wear due to service. Therefore it is understood that up to a mass imbalance of 120 Kg is allowed from field experiments. Mass imbalance can be created by either removing or adding mass. In this project to maintain the structural integrity of the drill string, a mass of 30 Kg has been added eccentrically to one of the sides of the drill collar. This is $\frac{1}{4}$ th of the mass imbalance that can be achieved without compromising the integrity of the drill string. The figure below shows how the mass imbalance has been created in the middle of the drill collar

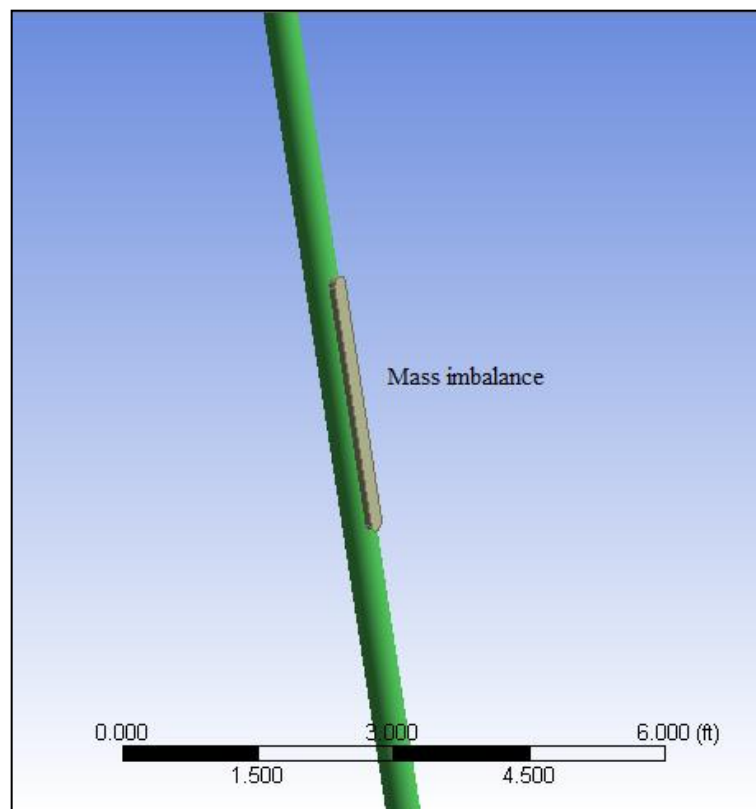


Figure 9: Mass imbalance in drill string

3.3.3 Materials and methodology

In this project, the drill string is modelled by SOLID73 element, which is used in three dimensional modelling of solid structures. This element is defined as having eight nodes with six degrees of freedom at every node. Each node is capable of translations in the x, y, z directions and rotations about the x, y, and z directions. This element is most suitable for modelling solid structures and requires less solution time. The contact between the elements is defined by Conta176 and Targe170 elements.

3.3.4 Boundary conditions

Since this project focuses on the lateral vibrations of drill string, the boundary conditions are set to observe lateral vibrations specifically. The wellbore effect will be included in the boundary conditions by allocating cylindrical supports at the place of the bit, stabilizers and rotary table. The radial supports are such that they can allow or constraint radial deflection and rotation about the drill string axis. They are specified as follows:

1. At the bit position, the drill string is allowed to rotate about its axis and allowed deflection in the radial direction
2. Since the purpose of the stabilizer is to restrict radial motion of the drill string, it is constrained radially but allowed to rotate about its axis
3. At the place of the rotary table, radial deflection of the drill string is constrained but it is allowed to rotate about its axis

The following figures shows the place of supports in the model

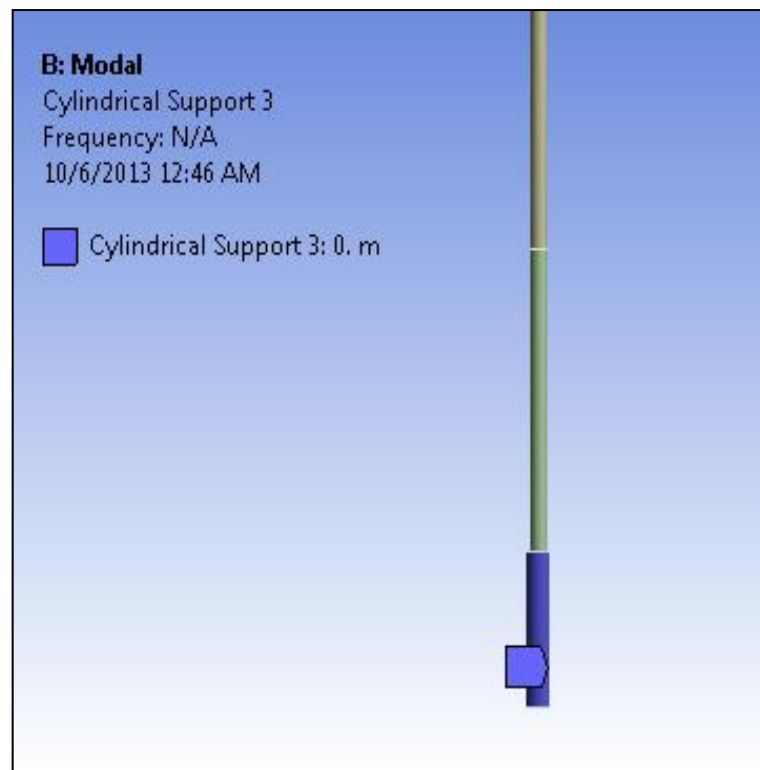


Figure 10: Cylindrical support at bit

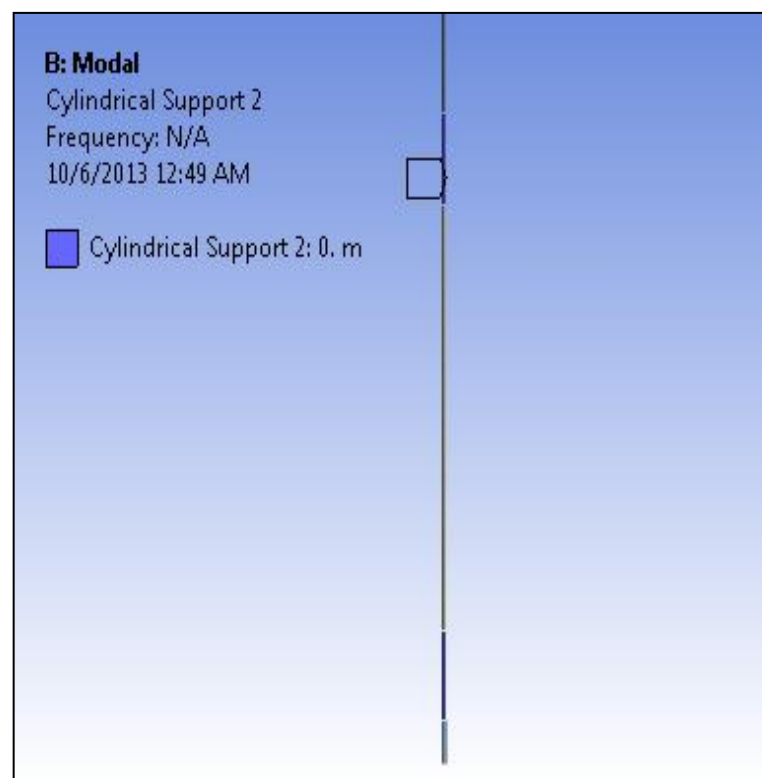


Figure 11: Cylindrical support at stabilizer

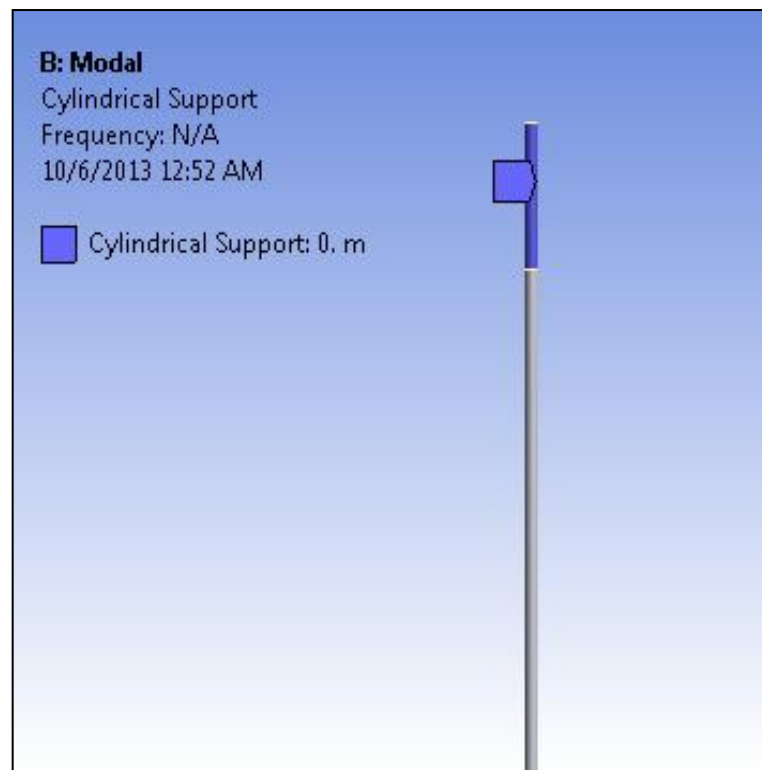


Figure 12: Cylindrical support at the Rotary table

3.4 Modal analysis

Modal analysis is used to determine a structure's vibration characteristics, its natural frequencies and mode shapes. As the problems are too complex to be solved analytically especially with the case of mass imbalance, finite element method is used. The types of equations which are observed in solving for modal analysis are those that are seen in the Eigen systems. A summary of the analysis performed on the model is explained in the appendix.

Finite element method procedure for modal analysis is explained below:

1. Geometry and Meshing: The geometry of the model is prepared from the drill string configuration as stated previously. Auto meshing is done to observe complex mode shapes
2. Analysis options: This represents the mode extraction options available to the user. Block Lanczos method is used in the case of this project. The number of modes to be extracted is specified as three.
3. Loading : The loading and constraints are done as specified in the boundary condition excluding mud effect in case of modal analysis
4. Solution: By clicking on the solve option, the software solves the equation of motion related to the geometry and produces the first three critical frequency and mode shapes

The validation of the result is confirmed by running the base case and comparing it with the experimental result. The next step is performing modal analysis on different drill string configurations along with their respective mass imbalance.

3.5 Harmonic analysis

Harmonic response analysis is used to determine the steady-state response of a linear structure to loads that vary sinusoidally or harmonically with time. Thus it is used to determine whether the structural design can overcome resonance, fatigue, and other harmful effects of forced vibrations.

Essentially there are two ways of performing harmonic analysis on the drill string. They are:

1. Full method – in which the harmonic analysis is done through a direct solution of equation of motion
2. Mode superposition – in which the harmonic response is obtained for a predefined load condition already solved for modal analysis.

Since modal analysis is already done, it is of advantage to use mode superposition method, where the linear combinations of Eigen vectors are already solved from modal analysis.

The procedure for performing harmonic analysis involves the following steps:

Create Analysis System

A modal analysis is a pre requisite for the harmonic analysis wherein, the engineering data, geometry and boundary conditions are shared

Define initial conditions

The number of modes extracted from modal analysis is 3 and the maximum frequency encountered for drill string 1 is 4.68 Hz. A general rule of thumb for specifying frequency range in harmonic analysis is to specify 1.5 times the maximum frequency encountered in the solution. Therefore the operating range of frequency for the analysis is from 0 to 7 Hz.

Apply Loads and Supports

The loading takes into account the effect of mud imposed on the drill string. The weight of 4049 lbs drill string in 12 ppg mud, from table 15(HWU, 2005):

$$= 4049 \times .817 \text{ (buoyancy factor)}$$

$$= 3308 \text{ lbs}$$

Therefore, 740 lbs of force acts as a tensile force on the drill string rotary table. Thus harmonic response for the second drill collar is done for the specified range of operating frequency.

Results

Since the second drill collar contains the mass imbalance in the drill string, harmonic response at this point is done to study the effect of displacements. The solution from this analysis is displayed as lateral displacements vs. frequency for an axially imposed load in the location of the rotary table

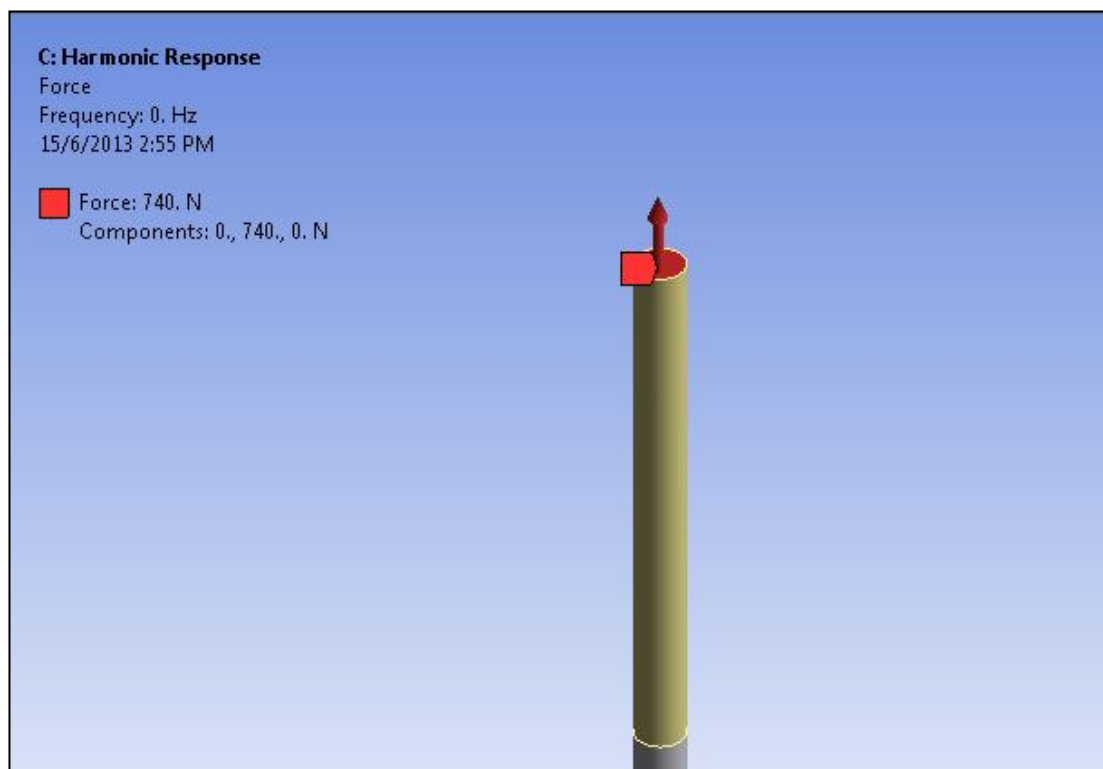


Figure 13: Harmonic analysis

CHAPTER 4

RESULTS AND DISCUSSION

4.1 Modal analysis

The aim of the modal analysis is to determine the natural frequencies and mode shapes. Comparison of simulation with that conducted by (Burgess) shows similarity in the results

The result for first three critical frequencies:

Table 7: Comparison of ANSYS result with Burgess Result

	length	1 st natural frequency	2 nd natural frequency	3 rd natural frequency
ANSYS	110(ft)	1.43	1.43	4.38
Burges et al(1987)	114(ft)	1.23	3.15	4.19

Natural frequencies of all the three drill string design along with their mass imbalance are summarized below:

Table 8: Natural frequencies of drill string under different conditions

Configuration	1st Natural frequency	2nd natural frequency	3rd natural frequency
Drill string 1	1.43	1.43	4.38
imbalance	2.03	2.23	4.63
Drill string 2	.62	.62	1.85
imbalance	.85	.90	1.83
Drill string 3	.36	.36	1.03
imbalance	.44	.54	1.06

The reason why 1st and 2nd natural frequency are equal is because of the symmetrical geometry (neglecting well inclination) used in simulation in which case the bending stiffness is equal about the strong and weak axis.

If the applied frequency or rotary speed matches with this natural frequency, resonance occurs and the amplitude of lateral vibration exceeds drastically. Thus the drill string collides with the wellbore and produces huge shocks. To prevent this from happening the operating speed or frequency must be out of these critical frequencies.

Since the normal operating speeds of a drill string rotary table is around 100-200 RPM, these critical frequencies are of important value. From the table, for drill string 1 it is clear that without mass imbalance, the resonance of the drill string occurs at 85 and 262 RPM which is clearly out of the operating speeds. But when a mass imbalance is introduced in the drill string, the resonance occurs at 121, 133 and 277 RPM. Thus two operating speeds cause resonance which can critically damage the drill string.

The different length of the drill string has a significant effect on the natural frequency and mode shapes. The natural frequency reduces as the length of the drill collar increases.

The mode shapes of drill string 1 for the first three deformations, subject to mass imbalance are shown in figures below:

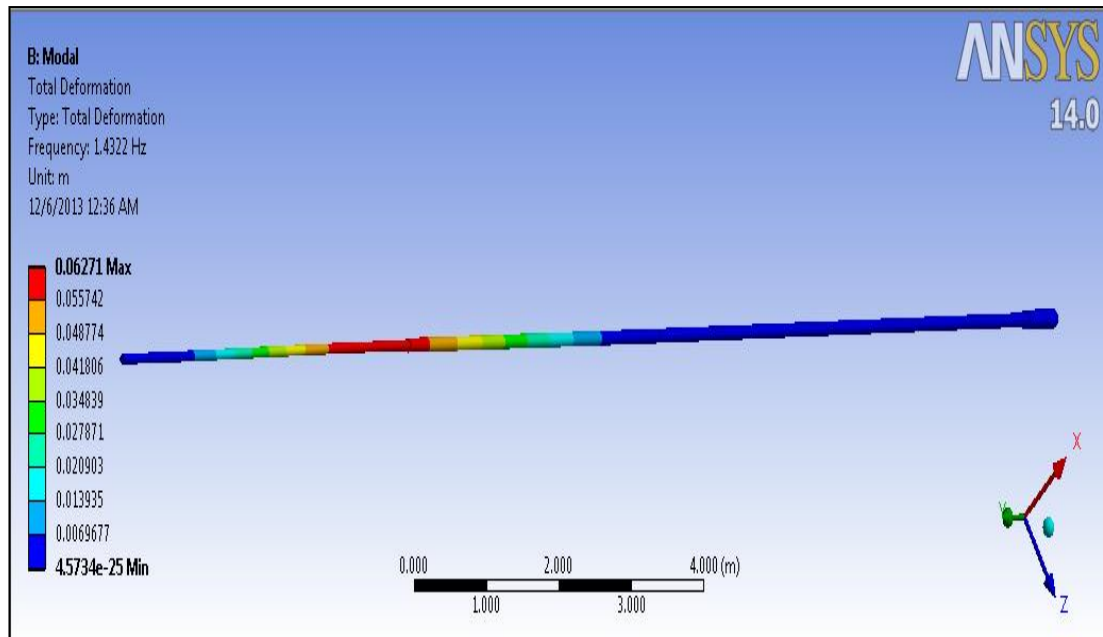


Figure 14: First deformation Drill string 1

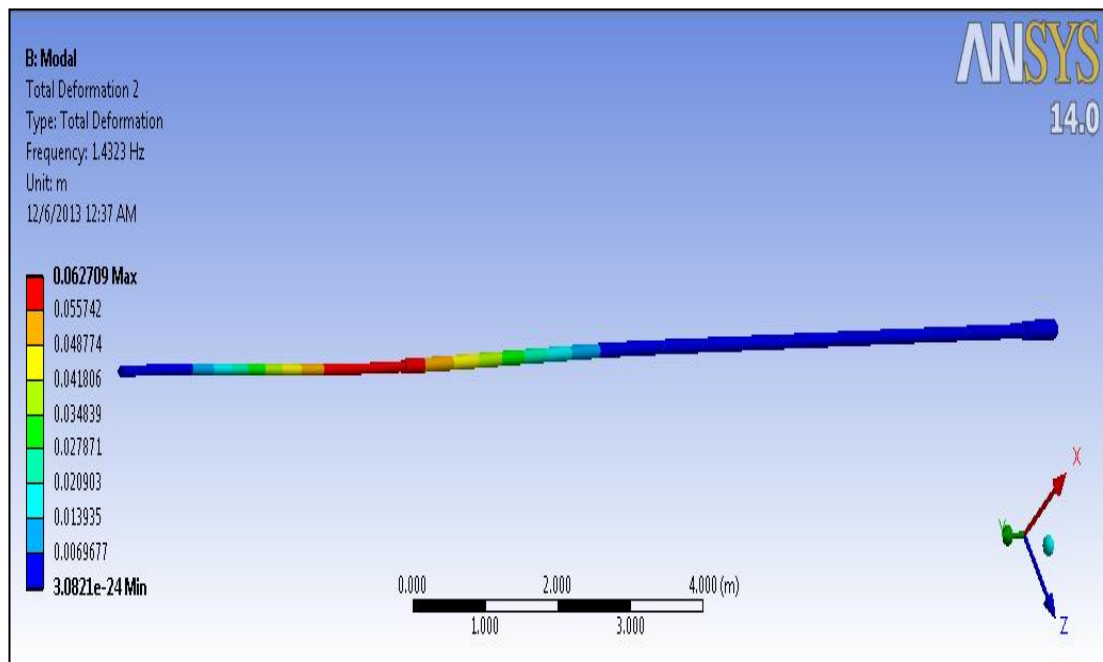


Figure 15: Second deformation Drill string 1

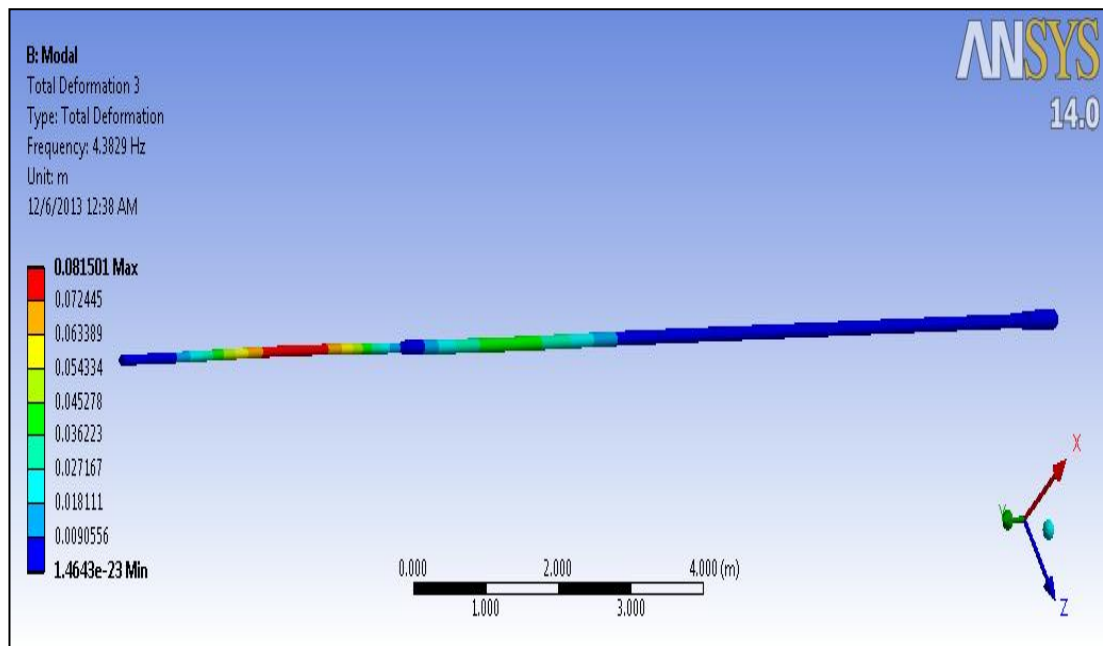


Figure 16: Third deformation Drill string 1

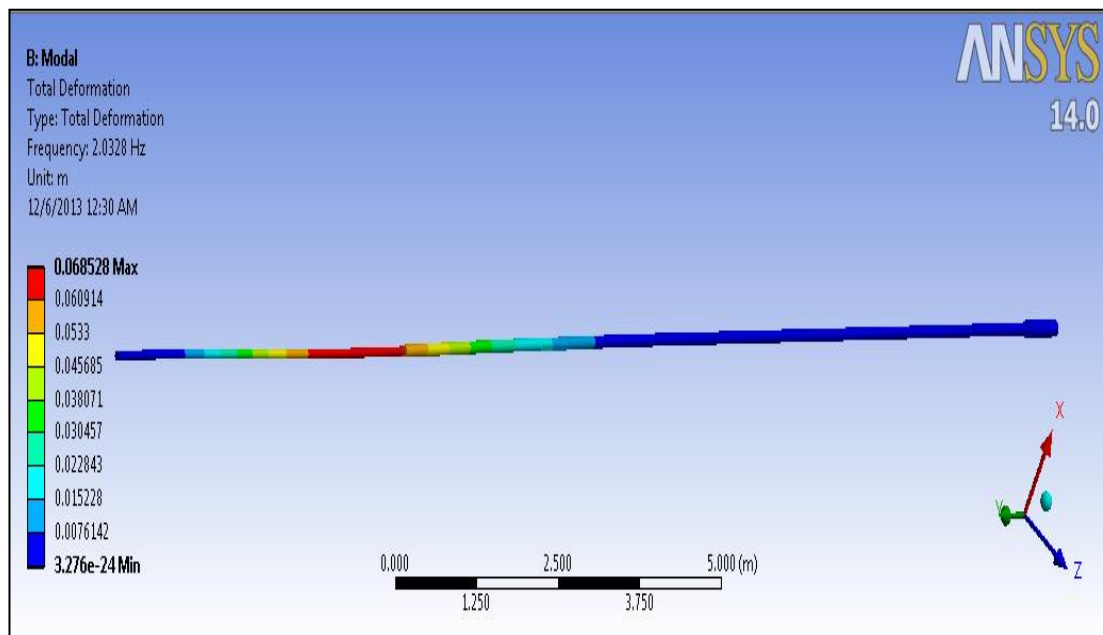


Figure 17: First deformation Drill string 1 with mass imbalance

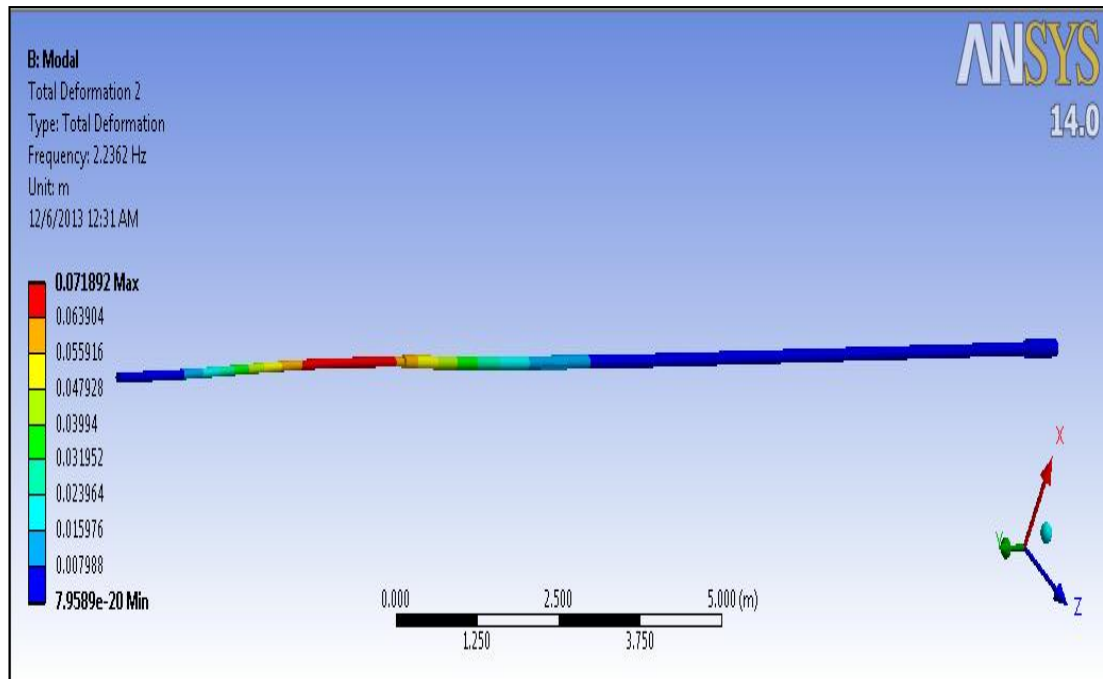


Figure 18: Second deformation Drill string 1 with mass imbalance

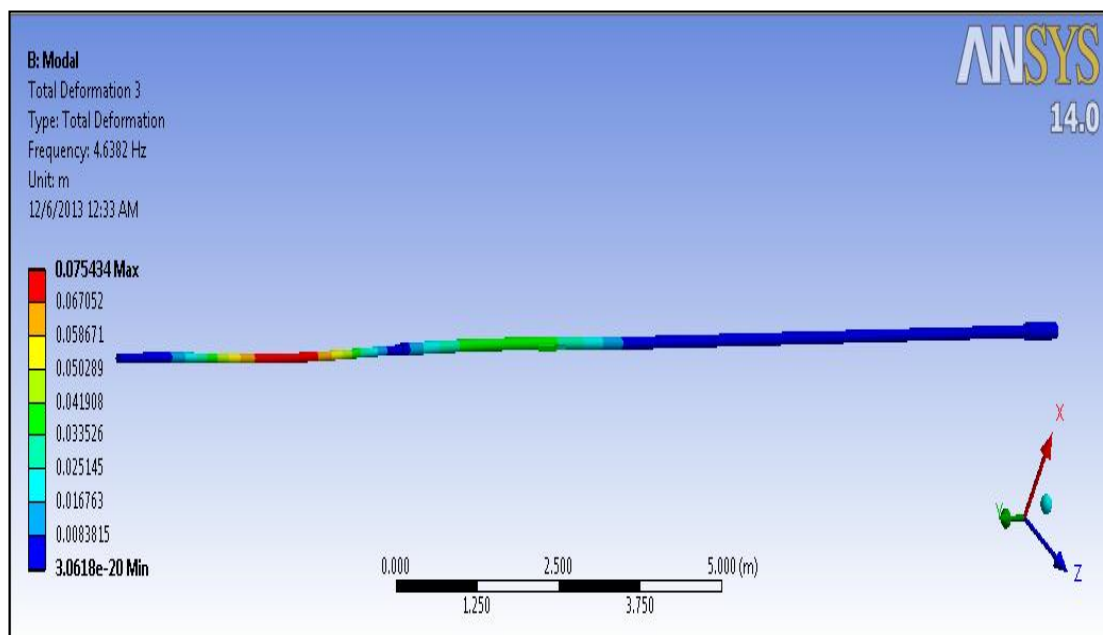


Figure 19: Third deformation Drill string 1 with mass imbalance

From the mode shapes, it is clear which areas of the drill string are subject to displacements. The lower part of the drill string which consists of the stabilisers and bit are subject to almost no displacements at all. This is due to the fact that the stabilisers negate the effect of displacements by constraining the drill string to only axial rotations without allowing any radial displacements. It is found that the maximum displacements are found at the place of the second drill collar which is well above the two stabilizers. The effect of mass imbalance can be seen on figure wherein the amplitude of displacements increases with the same configuration.

4.2 Harmonic analysis

Harmonic analysis is done mainly to understand the frequency response of a component of drill string when subject to a sinusoidal load. The critical component in this case is the second drill collar which is subject to mass imbalance. Therefore frequency response of the drill collar is plotted with respect to lateral deflections in the X direction.

It should be noted that while interpreting the normalized amplitude versus the frequency plot, the driving force at the bit is known until down hole measurements of vibration are available. The driving force will not be the same across every component of the drill string BHA. Additionally, the damping effect of the mud is not accounted for in the simulation, therefore it is best to interpret the peaks of the curve rather than the magnitude of displacements.

The following figures show the frequency response of the drill collar in terms of lateral displacements.

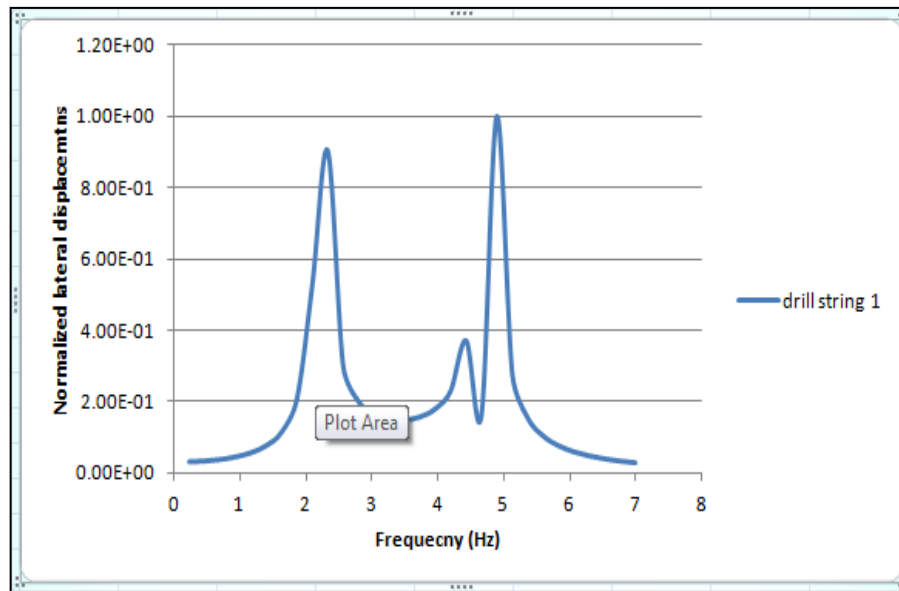


Figure 20: Harmonic analysis of drill string 1

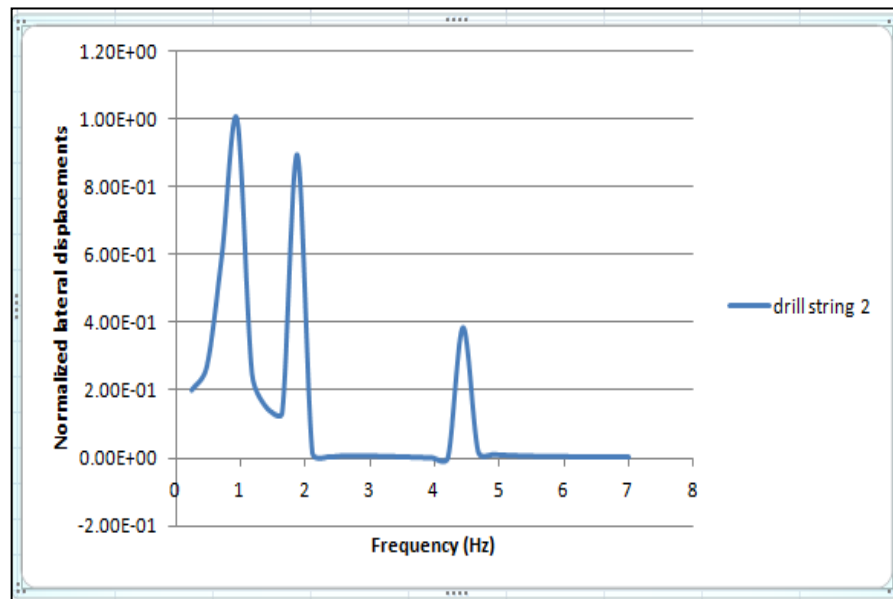


Figure 21: Harmonic analysis of drill string 2

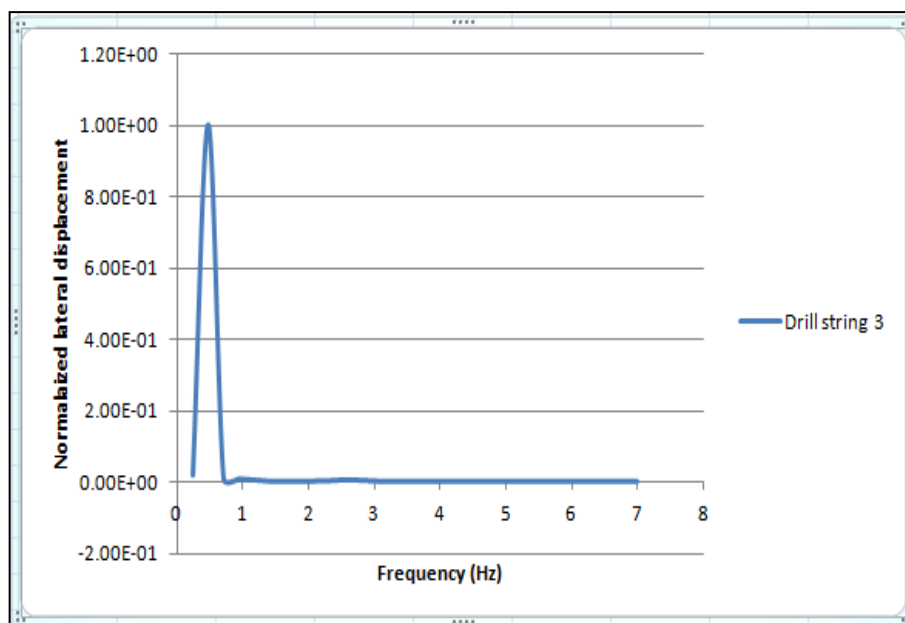


Figure 22: Harmonic analysis of drill string 3

The onset of resonance is indicated by the peaks in the harmonic response plot. This plot can be used to interpret which parts of the drill string is subject to large lateral displacements. In case of drill string 1, the drill collar is subject to large displacements at 4.8 Hz. This indicates that, the drill string is subject to maximum deflection at 4.8 Hz, while the peaks 2.2 and 4.2 Hz account for 90% and 38 % of maximum deflection. Thus, the operator has an idea of which frequency to avoid and which frequency to operate the drill string. Even though drill string experiences resonance at 4.2 Hz, the deflection of the drill string is not as severe as that encountered at 2.2 and 4.9 Hz. Similarly for drill string 2, the displacement is high at .8 and 1.9 Hz but only 40% of maximum displacement is experienced at 4.5 Hz. However, for drill string 3, maximum displacement occurs at .5 Hz, and very small Displacement occurs at 1 Hz, therefore the drill string is safe to operate at its usual operating conditions.

CHAPTER 5

CONCLUSION AND RECOMMENDATION

5.1 Conclusion

Lateral vibrations can cause a significant amount of failures in MWD tools, drilling tools and drill collars. This vibration effect is further amplified by the mass imbalance created by the MWD tools or the initial drill collar sag. The shocks produced by lateral vibrations can be higher than those which result from torsional or axial vibrations. This is because under lateral vibrations, the drill string collides with the wellbore wall, creating huge shocks.

The problem with lateral vibrations is that, it cannot be measured reliably from surface measurement unless it is a shallow well. Therefore in this project, a dynamic finite element model of a drill string was set up to conduct vibration studies. Modal analysis and Harmonic analysis were performed to identify the critical frequencies, mode shapes and frequency response of the drill string. The results of this project are very useful in understanding the effect of mass imbalance on drill string lateral vibration. This result can be used to identify the safe operating ranges of rotary speed for the drill string and identify the lateral displacement of a critical component for a range of frequencies.

As a rule of thumb, it can be understood that the more the mass and longer the drill string, the lower is the lateral resonant frequency. When compared to field experiment data, the ANSYS model has produced very close results; therefore it is very suitable for use of vibration studies by drilling engineers

5.2 Recommendation

It is highly recommended that

- Mass imbalance produced by MWD tools should be modelled more accurately with a fine detailed Analysis model
- Separate studies should be carried out to find the effect of drill collar sag on lateral vibrations
- The model was based on simple configuration, the effect of hole deviation and mud damping should also be integrated in the model in order to understand its effect on vibrations
- Torsional and axial vibration should also be studied with its appropriate boundary conditions

REFERENCE

1. Heriot Watt University (HWU), 2005, "Drilling Engineering", Manual of Heriot Watt Institute of Petroleum Engineering, Edinburgh, UK
2. Kreuzer. E and Struck. H, 2003 "Mechanical Modelling of Drillstrings" In: PAMM - Proc. in Appl. Math. And Mech
3. P.R. Paslay, Yin-Min Jan, May 24, 1994, "Methods of Determining Drillstring Bottom Hole Assembly Vibrations", United States Patent, Patent Number 5313829,
4. Dawson,R., Lin, Y.Q., and Spanos, June 14-19 1995: "Drill string Stick-slip oscillations", S.P.E, Houston, United States of America.
5. R.I. Leine, September 1997, "Literature Survey on Torsional Drill String Vibrations", Eindhoven University of Technology, Netherlands.
6. R. I. Leine, D. H. van Campen, 2000, "Stick slip whirl interaction in drill string dynamics", Eindhoven University of Technology, Netherlands.
7. Schlumberger, 2010, "Drill string vibrations and vibration modelling", Houston, United States of America
8. P. Harvey and M Wassell, 1991, "The design of steerable BHAs to minimize the adverse effects of motor imbalance and drilling forces", SPE, Dallas, TX
9. M.W Dykstra, D.C Chen, T.M Warren, J.J Azar 1995, "drill string component mass imbalance: A major source of down hole vibrations", SPE, Drilling conference, Amsterdam
10. F. de S. M. Costa and P.R Ribeiro, 1997 "Finite element modelling of the mechanical behaviour of unbalanced drill collars" SPE, Rio de Janeiro conference
11. Jamal Zare, Seyed Jalalodin Hashemi and Gholamreza Rashed, 2011, "Finite element analysis of drill string lateral vibration", Petroleum university of technology, Iran
12. T.M Burgess, G.L Mc Daniel and P.K Das, 1987,"Improving BHA tool reliability with drillstring vibration models: Field experience and limitations", SPE, Drilling conference, New Orleans, Los Angeles, U.S.A

APPENDIX

1. For Modal analysis, the ANSYS software solves the matrix equation of motion for the free vibration of the structure

$$[M]\{\ddot{u}\} + [K]\{u\} = 0$$

The solution to the above equation gives the natural frequencies (Eigen values) and mode shapes (Eigen vectors).

2. For harmonic motions, replacing $\{\ddot{u}\} = -\omega^2\{u\}$ which gives the Eigen value expression:

$$[K]^{-1}[M]\{u\} = [I]\{u\}$$

Where $[I]$ is the unity matrix.